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A METHODOLOGY AND ESTIMATION OF FILM THICKNESS IN ELASTOHYDRODYNAMIC LUBRICATION

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ABSTRACT

Cam and tappetare always under gigantic stresses which lead to wear and tear of the same. Hence it must be identified, estimated and minimized for prolonged and smooth running of engine. This paper describes an approach to estimate wear prediction in cam and tappet of diesel engines through force and wear analysis and which leads to estimation of minimum film thickness between cam and tappet. So that wear rate can be reduced up to certain extent. Through inertia forces accurate valuation of contact stresses has been done which imparts vital role in prediction of wear. Contacting loads and relative motions have been deduced from kinematic and dynamic analysis of valve gear train. Estimation of wear has been done by using Archard's wear model. For film thickness evaluation, Dowson-Hamrock's curve fitting formula has been used considering elastohydrodynamic lubrication. In this analysis, the flat-faced tappet has been used with Polydyne cam profile.

KEYWORDS: Gigantic, Dowson-Hamrock's Curve & Polydyne

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Original Article

INTRODUCTION

The functioning of valve train controls the opening and closing of valve timing which is affected by mechanical and dynamic factors of the engine. This directly affects the work done, torque and emission of exhaust gases [1]. Sub surface fatigue may evolve due to peak accelerations in the engine which causes higher Hertz pressures [6]. At lower speeds, surface wear predominates due to weakening of lubricant layer and higher friction between cam and tappet [4, 5, 7, 10, and 11].Grievous tribological conditions evolve in cam-tappet systems due to fluctuating loads and abrupt variation of relative velocities. The forces for opening of valves in gear train are extremely affected by stiffness and damping features of valve spring and valve seat, masses and geometry of components and frictional behavior of contacting components.

The non-conformal geometry of cam and tappet comes up with abysmal fluid film pressures and high-stresses in the contact region. Hence, it gives rise to the inclusion of elastic deformation, occurring in contact region, in the computation of lubricant film thickness, through Elastohydrodynamic lubrication (EHL) analysis. These parameters account for non-linear kinematics and dynamics of valve train system. Both shape and thickness of fluid-film between cam and tappet depend on the applied load and relative speed [12, 13, and 14]. In the present work, a flat faced follower is used polynomial motion of 5^{th} or 6^{th}-order, which gives better performance at high speeds[2]. However the analysis can easily be extended to other shapes of tappet.

Through optical interferometer [16], lubricant film thickness can be obtained which uses monochromatic and dichromatic light sources of two different wavelengths. Also ultrasound measurement technique [15] can be used for the same which utilized the reflection of ultrasound to measure the lubricant film thickness.

THE CAM PROFILE

The cam profile, used in this model, is a Polydyne [2 and 3] of 6-order, the equation of motion for this cam is:

Lift, $$Y = L\left[64\left(\frac{\theta}{\beta}\right)^3 - 192\left(\frac{\theta}{\beta}\right)^4 + 192\left(\frac{\theta}{\beta}\right)^5 - 64\left(\frac{\theta}{\beta}\right)^6\right] \quad (0 \leq \theta \leq \beta) \tag{1}$$

Velocity, $$V = L\omega\left[192\left(\frac{\theta^2}{\beta^3}\right) - 768\left(\frac{\theta^3}{\beta^4}\right) + 960\left(\frac{\theta^4}{\beta^5}\right) - 384\left(\frac{\theta^5}{\beta^6}\right)\right] \tag{2}$$

Acceleration, $$A = L\omega^2\left[384\left(\frac{\theta}{\beta^3}\right) - 2304\left(\frac{\theta^2}{\beta^4}\right) + 3840\left(\frac{\theta^3}{\beta^5}\right) - 1920\left(\frac{\theta^4}{\beta^6}\right)\right] \tag{3}$$

Jerk, $$J = L\omega^3\left[\frac{768}{\beta^3} - 4608\left(\frac{\theta}{\beta^4}\right) + 11520\left(\frac{\theta^2}{\beta^5}\right) - 7680\left(\frac{\theta^3}{\beta^6}\right)\right] \tag{4}$$

Where $\theta = \omega \times t$

ω =Angular velocity of camshaft = $\frac{2\pi N}{60}$ rad/s

N = Rpm

β = Constant (max. angle of rise or fall)

ELASTOHYDRODYNAMIC LUBRICATION (EHL)

At extreme loads and higher speeds, it is much difficult to maintain an oil film with a sufficient thickness for normal hydrodynamic lubrication. In these special cases, the lubricant is compressed and extreme high pressures are developed. The lubricant's viscosity increases at these high pressures and elastic deformation of metals take place. This is termed as Elastohydrodynamic lubrication. In this realm of lubrication, an increase in load deforms the metal surfaces rather than affecting the oil film thickness, as the oil film is actually more rigid than the metal. Such type of lubrication exists in rolling contact bearings, gears, cam-tapet, etc.

PRESSURE DISTRIBUTION IN ELASTOHYDRODYNAMIC CONTACTS

The profile of pressure distribution [7] is hemispherical or ellipsoidal according to classical Hertzian theory under static load condition. The pressure field changes when the surfaces start moving relative to each other in the presence of a piezoviscous lubricant such as oil. The pressure profile changes greatly at the entry and exit regions of the contact. Subsequently, at the entry region, the value of hydrodynamic pressure is lower than for a dry Hertzian contact. The characteristics of the pressure peak depend largely on the lubricant's pressure-viscosity characteristics. The pressure distribution is shown in the given figure [2].

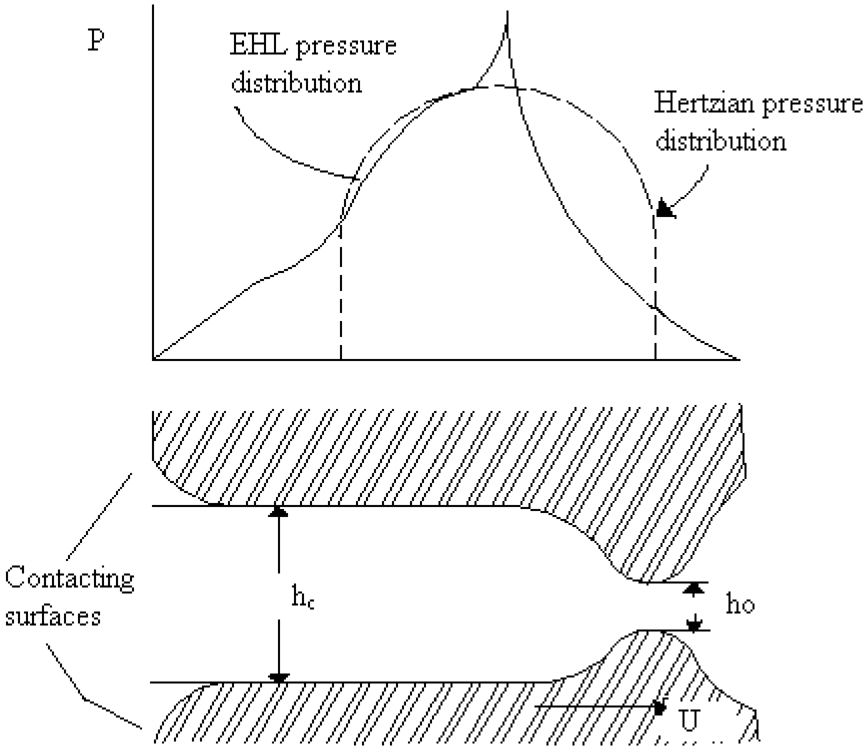

Figure 2: Hydrodynamic pressure distribution in an Elastohydrodynamic Contact

Where h_c is the central film thickness, h_o is the min. film thickness

In order to calculate the Hertz pressures, the theory of rolling [6] contacts has been used.

$$P_H = 0.418\sqrt{\frac{F_1 E'}{b_n}\left(\frac{1}{R_p} + \frac{1}{RON}\right)} \tag{5}$$

Where $RON = \left(h_p + r_o + \frac{a_p}{\omega^2}\right)$

$$\frac{1}{E'} = 0.5\left(\frac{1-\nu_1^{\ 2}}{E_1} + \frac{1-\nu_2^{\ 2}}{E_2}\right) \tag{6}$$

LUBRICATION MODELLING

The following assumptions are made in the calculation of minimum film thickness calculation for cam and tappet contact:

- The displacements are calculated for a semi-infinite solid in a condition of plane strain.
- Side leakage is neglected.
- The boundary conditions for pressure are: at inlet p=0 at a large distance from the high-pressure zone; at outlet p= ∂p/∂x =0.
- The lubricant is incompressible.

- Thermal effects are neglected.

Dynamic Viscosity of Lubricant (η)

Lubricant used is SAE-30.

Specific gravity = 0.885

Dynamic viscosity =1.033 Poise

The imposed variables are:

R = effective radius of the sliding pairs.

$$= \frac{R_p \times RON}{R_p + RON}$$

Where R_p = Radius of tappet

RON = Radius of curvature of cam

E' = Effective elastic modulus of the sliding pair

ν_1, ν_2 = Poisson's ratio of cam and tappet respectively

η_o = Viscosity of the lubricant at condition to entry contact

α = Pressure exponent of viscosity given by

$\eta = \eta_o \exp(\alpha p)$

w, u = External variables, load per unit width and speed

H_m = minimum film thickness at point of contact

The Load Parameter:

$$W = \frac{w}{E' \times R}$$

The speed Parameter:

$$U = \frac{\eta_o \times u}{E' \times R}$$

The Materials Parameter:

$$G = \alpha \times E'$$

According to Dowson-Hamrock's curve-fitting formula:

Minimum film thickness is given by

$$H_m = 1.714 \times R \times W^{-0.128} \times U^{0.694} \times G^{0.568} \tag{7}$$

Lubrication Analysis using equation (7), minimum film thicknesses for inlet and exhaust cam were calculated at top of the cam and 10 deg of camshaft angle. The whole calculation has been done in excel programming.

RESULTS

Minimum Film Thickness between Cam and Tappet

Minimum film thickness at top of the cam

Table 1

Speed(rpm)	Minimum Film Thickness (μm)	
	Inlet Cam	Exhaust Cam
3200	1.09	0.89

Minimum film thickness at 10^{o} of the camshaft angle

Table 2

Speed(rpm)	Minimum Film Thickness (μm)	
	Inlet Cam	Exhaust Cam
3200	1.11	0.91

CONCLUSIONS

The cam and tappet of IC Engines are under variable loading conditions, hence tribological parameters are usually difficult to evaluate. The design of valve gear train of I C engines is extremely intricate, leading to many problems while analyzing the contact forces. For cam materials, the usual permissible Hertzian pressure is between 850 to 1000 MPa.

This paper has lubrication analysis based on Dowson Hamrock's curve fitting formula and obtained values of minimum film thickness at top of cam and at 10 degree of cam rotation are well within the usual values in case of cam and tappet elastohydrodynamic lubrication. In this paper, dynamic model has limitation of damping and vibration effects hence the prediction of wear at higher speeds (above 3200 rpm) may deviate considerably from the actual results, because this has not been included in the analysis. Further optical interferometer and ultrasound measurements can be used to get the film thickness of lubricants through experimentation.

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NOMENCLATURE

a_p = Acceleration of Tappet

b_n = Cam width

h_p = Follower lift

R_o=Base circle radius

R_p = Radius of Follower

W= Load

R = Effective radius of the sliding pairs

RON = Radius of curvature of cam

E' = Effective elastic modulus of the sliding pair

E_1= Elastic modulus of cam

E_2= Elastic modulus of tappet

$\nu_{1,}\nu_2$ = Poisson's ratio of cam and tappet respectively

η_o = Viscosity of the lubricant at condition to entry contact

η=Dynamic viscosity of lubricant

α = Pressure exponent of viscosity given by, $\eta = \eta_o \exp(\alpha P)$

w,u = External variables, load per unit width and speed